



THE MECHANISMS AND SCALING OF DAMPING IN A PRACTICAL STRUCTURAL JOINT

By Brantley R. Hanks and David G. Stephens

NASA Langley Research Center
Langley Station, Hampton, Va.

Presented at the 36th Shock and Vibration Symposium

FACILITY FORM 602	N 68-27545	
	(ACCESSION NUMBER)	(THRU)
	26	1
	(PAGES)	(CODE)
	1MX-59959	32
	(NASA CR OR TMX OR AD NUMBER)	(CATEGORY)

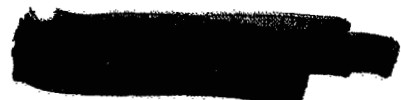
Los Angeles, California
October 18-20, 1966

GPO PRICE \$ _____

CFSTI PRICE(S) \$ _____

Hard copy (HC) _

Microfiche (MF) _



THE MECHANISMS AND SCALING OF DAMPING IN A PRACTICAL STRUCTURAL JOINT


Brantley R. Hanks and David G. Stephens
NASA Langley Research Center
Langley Station, Hampton, Va.

ABSTRACT

An investigation was conducted to determine the effect of geometric scale on the damping in a practical beam-joint assembly. A cantilever configuration was utilized wherein the beam was bolted between two angle brackets at the support. Four geometrically similar assemblies, covering a scale range of approximately twenty to one, were tested. Free decay of the fundamental mode was measured over a range of joint clamping pressures and beam tip amplitudes. Also, damping changes resulting from the addition of liquid lubricants and viscoelastic films to the joint interfaces were investigated. Data indicate that an increase in model size results in a decrease in damping attributed to the structural joint. Furthermore, joint damping is shown to be slightly dependent on vibration amplitude and to vary as an inverse function of joint clamping pressure. Joint damping may be substantially increased by the addition of liquid lubricants or viscoelastic films at the joint interfaces.

INTRODUCTION

Dynamic models are often used to study the vibratory response of complex systems when full-scale testing is precluded by system size and/or cost. The usefulness of model tests is dependent upon a knowledge of the proper scaling relationships required to extrapolate model data to the full-scale systems.

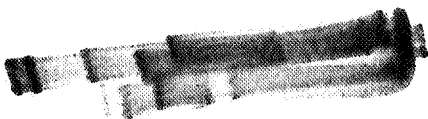


A considerable amount of information has been obtained on the scaling relationships for frequencies and mode shapes. However, the variation of damping with model size or scale is largely unknown and often either neglected or considered to be the same in both model and prototype. The development of proper scaling relationships for damping requires a knowledge of each of the damping mechanisms in the system such as material hysteresis (1, 2), air damping (3, 4), and joint damping (5, 6, 7), for example. In space systems, scaling relationships for joint damping are of particular importance since the major source of energy dissipation in such systems (8) is usually attributed to structural interfaces or joints.

The purpose of this paper is to present the results of an experimental investigation of the nature and scaling of damping in a structural joint. The joint damping of four cantilever systems, covering a geometric scale range of twenty to one, was examined. Data are presented to show the effect of vibration amplitude, joint clamping pressure, and model scale, as well as the effect of interface lubricants and films on the magnitude of the damping in structural joints.

APPARATUS AND TEST PROCEDURE

The apparatus used in this investigation is shown in figures 1 and 2. It consisted of four geometrically scaled beams bolted between correspondingly scaled angle brackets. The angle brackets were, in turn, clamped to a massive concrete and steel supporting block giving a cantilever beam configuration. The four models had scale factors, λ , of 1, .667, .333, and .053 with the beams



ranging from 5 feet down to 3.2 inches in overall length. The beams and angle brackets were made of 6061 aluminum alloy with all surfaces finished to 63 micro-inches rms. Each angle bracket was machined from a single piece of aluminum and had a web welded to the center to provide rigidity.

Three joint interface conditions were studied (dry, oil-coated, and viscoelastic-film-filled) in an effort to find a method for improving the inherent damping of both small and large scale systems. The dry and oil-coated joint tests were performed on all four models whereas the effect of viscoelastic films was studied on the .667 assembly. In studying the effect of joint lubrication, the interfaces were coated with a thin layer of oil before assembly. Three oils having viscosities of 150, 525, and 1400 centipoises were used. The effect of viscoelastic film inserts was studied using three film materials - .5 mil Teflon, 1 mil Mylar, and 1 mil polypropylene. The films were cut to the shape of the joint interface and placed in position during model assembly.

The test procedure was essentially the same in all cases. The total damping of the systems was measured at atmospheric pressure for a range of joint clamping pressures (by varying bolt tightening torque) and beam tip amplitudes. For a particular clamping pressure, the beam was deflected manually and released to oscillate in the first cantilever vibration mode. Oscillations of the beam were sensed by an electrical resistance strain gage attached to one side of the beam as shown in figure 1. The strain gage was coupled, through an amplifier, to an electronic dampometer which is a device for determining the frequency and damping of a vibrating system. Basically, the dampometer counted the number of cycles as the amplitude decayed between preset limits. The logarithmic decrement, δ ,

was then calculated from the equation

$$\delta = \frac{1}{N} \log_e \frac{y_n}{y_n + N} ,$$

where N was the number of cycles counted, y_n was the amplitude at which counting started, and $y_n + N$ was the amplitude at which counting ceased. In all tests the ratio of start to stop amplitude was maintained at 10/7 so that

$$\delta = \frac{1}{N} \log_e \frac{10}{7} .$$

Since the damping was measured over a band $[y_n, y_n + N]$ of the decay envelope, the logarithmic decrement was specified at the average amplitude of this band. Measurements were made at several amplitude levels for each bolt torque by varying the triggering voltage of the dampometer. In all tests, sufficient initial deflection was given to the beam to allow transients to die out before the dampometer triggering amplitude was reached. Each test was repeated at least five times and the average value of the data was used in analysis.

Joint clamping pressures were calculated from the bolt tightening torques using the formula (9)

$$F = \frac{T}{0.2D} ,$$

where F is the clamping load per bolt, T is the torque, and D is the bolt diameter. Converting to average clamping pressure produced on the joint by the four bolts, the formula becomes

$$P = \frac{20T}{DA},$$

where P is the average clamping pressure and A is the joint interface area. No allowance was made for variation of clamping pressure across the interface.

MATERIAL DAMPING CONSIDERATION

The determination of the magnitude of joint damping in a complex system involves the separation of the total damping into its various components. One contribution to the total damping is that of material or hysteresis damping within the material comprising the system. Experimental separation and measurement of this material damping is difficult in complex systems such as the one under study. However, an analytical expression developed by Zener (1) has been verified for aluminum in experimental work by Granick and Stern (2). Material damping in a cantilever beam was shown to be closely approximated by the equation

$$\delta_m = \frac{\alpha^2 ET}{\pi c} \left[\frac{\omega \gamma}{1 + \omega^2 \gamma^2} \right],$$

where

δ_m = logarithmic decrement for material damping,

α = thermal coefficient of linear expansion, $1/^\circ R$,

E = modulus of elasticity, psi,

T = absolute temperature, $^\circ R$,

c = specific heat per unit volume, $BTU/in.^3-^\circ R$,

ω = circular frequency of vibration, rad/sec.

and for a flat beam of uniform thickness

$$\gamma = \frac{t^2 c}{\pi^2 K},$$

where

t = beam thickness, in.,

K = thermal conductivity, BTU/sec-°R-in.

The material damping as predicted by this equation for the four beams tested in this joint damping study is shown as a function of frequency in figure 3 on the left. The magnitude of the material damping at the fundamental resonant frequency is denoted for each beam by a circular symbol. These resonant damping values are replotted as a function of scale factor on the right in figure 3. For the systems under study, material damping as predicted by the Zener equation is essentially inversely proportional to scale.

PRESENTATION AND DISCUSSION OF RESULTS

The test program consisted of an isolation and examination of the damping for the variables: vibratory amplitude, joint clamping pressure, model scale, and interface condition (i.e., dry, lubricated, or film insert). The dependency of the damping on each of these variables is illustrated by representative data in the following sections.

Dry Interface

Effect of vibration amplitude.- The damping measured for the 0.667 scale model, which is typical of the four assemblies, is shown in figure 4. The total damping in terms of the log decrement, δ , is presented as a function of

the ratio of vibration displacement amplitude to beam thickness, y/t , for five values of joint clamping pressure. The total damping increases linearly with an increase in amplitude. Since the total damping represents not only losses in the joints but also internal hysteresis and air damping, the question arises as to whether the joint damping per se is amplitude dependent. Several factors suggest that the joint damping is amplitude dependent. First, references 2 and 3, respectively, indicate that both the hysteretic and air-damping losses are amplitude independent for the range of amplitudes covered by these tests. Secondly, the slope of the faired lines in figure 4 is observed to change with a change in clamping pressure (a variable affecting joint damping only) with all other factors being held constant.

Effect of joint clamping pressure.- The damping for each of the models is presented as a function of joint clamping pressure in figure 5. These curves are cross plots of the damping-amplitude curves such as the previous example, figure 4. The total damping decreases rapidly as the clamping pressure is increased in low range; however, it levels out or approaches a constant value at relatively high clamping pressures. It will be assumed henceforth that the joint damping at high stress levels is negligible compared to the other or tare damping (the damping due to the surrounding air and internal hysteresis) in the system. Thus, the magnitude of the joint damping at a particular pressure and amplitude is considered to be the difference between the measured value and the respective high clamping stress asymptote.

Effect of scale.- The variation of total damping with scale factor is shown in figure 6 for two values of the joint clamping pressure. All of the data between the amplitude limits, y/t , of 0.03 to 0.08 fall within the indicated band. These data demonstrate that total damping increases with decreasing scale

factor resulting in substantially higher damping for the smaller models. The trends of these curves are very similar to the variation of material damping with scale factor as predicted by Zener, which is repeated for comparison purposes.

The damping attributed to the joint, obtained by subtracting the respective high stress asymptote (fig. 5) from the above curves of total damping (fig. 6), is presented as a function of scale factor in figure 7. The joint damping is also seen to be inversely proportional to the scale factor. Examination of the curve reveals that the joint damping values for the 0.333 and 0.053 scale models are several times higher than those of the larger models. These curves indicate that caution should be used in extrapolating damping data obtained in tests of small models to full-scale systems. The common practice of assuming that the damping of the prototype is the same as in the model could lead to gross over-estimates of the damping in the full-scale systems.

Treated Interfaces

Effect of oil.— In an effort to alter the joint damping, the effect of interface lubricants were examined. Typical results are shown in figure 8 where the total damping for the 0.667 scale model, with 150 cp oil added to the joint interfaces, is presented as a function of joint clamping pressure. For comparison purposes the "dry" data (fig. 5) are repeated.

The dependency of the damping on clamping pressure is essentially the same in both the lubricated and the dry cases. However, the magnitude of the damping recorded for the lubricated joint is considerably higher. A similar phenomenon was reported in reference 10 where the damping of a cantilever beam was substantially increased by the addition of grease at the root although the exact mechanism was not fully discussed.

The dependency of the damping on amplitude, as indicated by the width of the band in figure 8, is higher for the lubricated case. This was noted for all models except the 0.333 scale model for which there was little difference in amplitude dependency between the dry and lubricated joints. Although data are not presented, the dependency of the damping on amplitude was again found to be linear. Also, tests conducted with the 525 and 1400 centipoise oils revealed no appreciable difference when compared with the damping in the joint lubricated with the 150 cp oil.

The relative effect of oil on damping for all four models is summarized in figure 9 where the ratio of total damping with oil to that without oil is plotted as a function of scale factor. A lubricant is shown to increase the damping in all but the smallest model where the addition of oil slightly decreased the damping.

Effect of viscoelastic films. - The effect of adding viscoelastic films to the joint interfaces of the .667 scale model is illustrated in figure 10. In this figure, the range of total damping obtained with each of the three film materials is shown together with those obtained in the dry and lubricated joint cases in bar graph form. The film materials are shown to substantially increase the damping although they are not significantly more effective than oil in this respect.

CONCLUSIONS

An investigation was conducted to determine the effect of geometric scale on the damping in the joint of a practical beam-joint assembly. In addition, a brief study was made of the effect on joint damping of adding oil and viscoelastic

films to the joint interfaces. Within the range of variables considered in these studies, the following conclusions were noted:

1. The damping attributed to the structural joint increases essentially linearly with vibration amplitude for any given joint clamping pressure.
2. Total damping decreases with increasing joint clamping pressure for low clamping pressures but becomes essentially invariant at high pressures.
3. Both the total damping of the assembly and the joint damping increase considerably with a decrease in geometric scale.
4. The addition of oil to joint interfaces can, depending on model size, increase damping over that of the dry joint case. Thin viscoelastic films inserted between joint interfaces are also effective in increasing damping but not significantly more so than oil.

REFERENCES

1. Zener, C. M.: Elasticity and Anelasticity of Metals. University of Chicago Press, Chicago, 1948.
2. Granick, Neal, and Stern, Jesse E.: Material Damping of Aluminum by a Resonant-Dwell Technique. NASA TN D-2893, August 1965.
3. Baker, W. E., and Allen, F. J.: The Damping of Transverse Vibrations of Thin Beams in Air. Ballistics Research Laboratories Report 1033, October 1957.
4. Stephens, David G., and Scavullo, Maurice A.: Investigation of Air Damping of Circular and Rectangular Plates, a Cylinder, and a Sphere. NASA TN D-1865, April 1965.
5. Ungar, Eric E.: Energy Dissipation at Structural Joints; Mechanisms and Magnitudes. Bolt Beranek and Newman, Inc. (Air Force Contract No. AF 33(657)-10125), August 1964.
6. Goodman, L. E., and Clumpp, J. H.: Analysis of Slip Damping with Reference to Turbine Blade Vibration. J. Applied Mech., vol. 23, September 1956, pp. 421-429.

7. Pian, T. H. H., and Hallowell, F. C., Jr.: Structural Damping in a Simple Built-up Beam. Proc. First U. S. National Congress of Appl. Mech., ASME, New York, 1952, pp. 97-102.
8. Osgood, Carl C.: Force Transmissibilities in Spacecraft Structures. The Shock and Vibration Bulletin, Bulletin 35, Part 7, April 1966, pp. 205-211.
9. Technical Information Staff of Industrial Fasteners Institute: Joint Design. Machine Design, Fasteners Book Issue, March 21, 1963, pp. 33-40.
10. Klint, R. V., and Owens, R. S.: The Effect of Root Lubrication on the Damping of Cantilever Beams. Paper presented at the Lubrication Conference (New York, New York), October 1959.

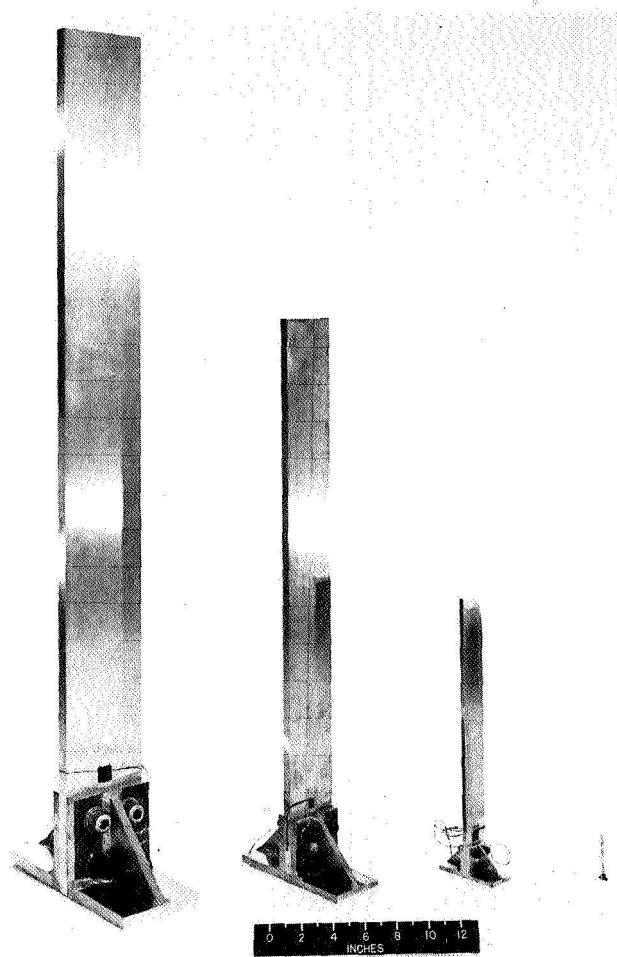
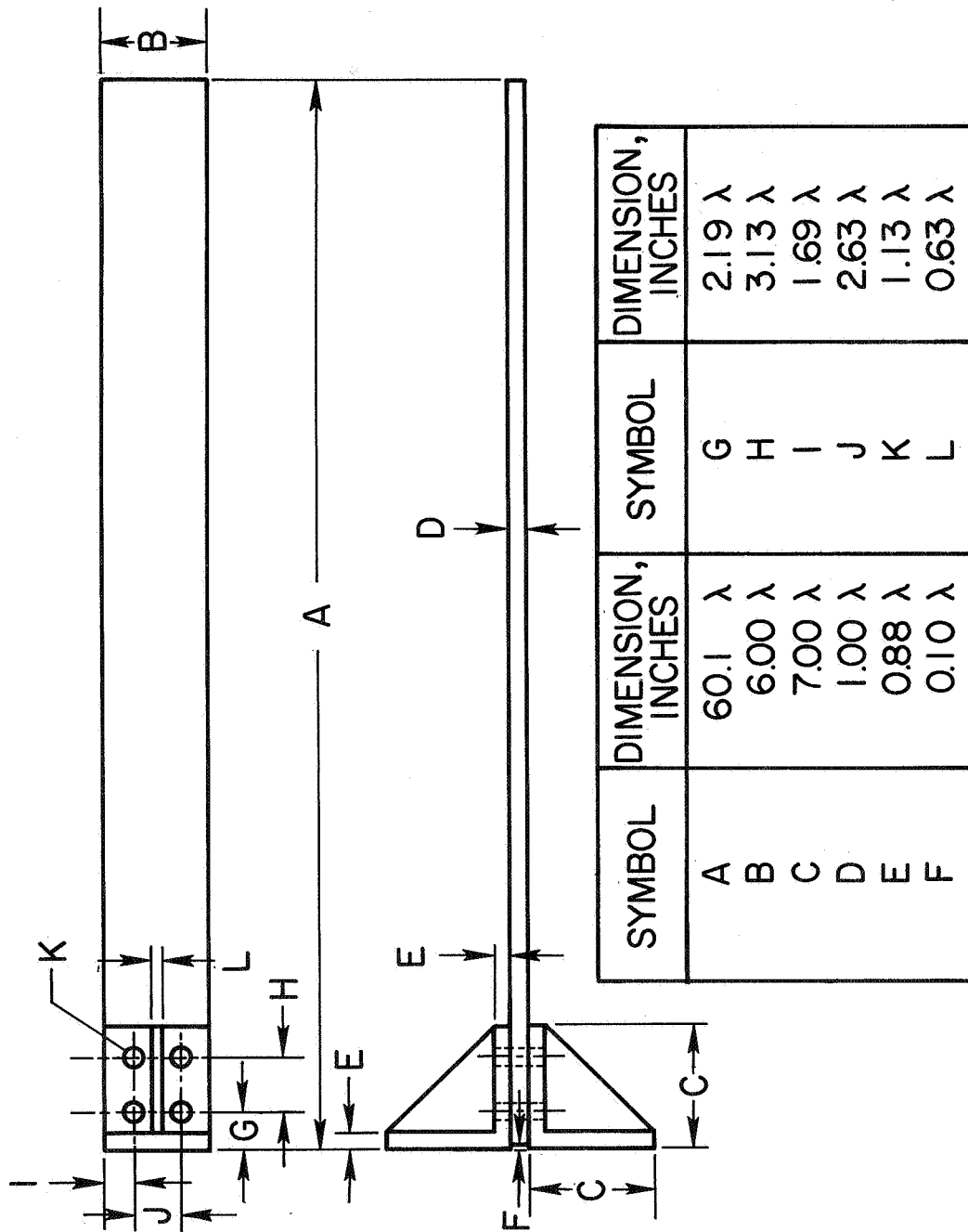


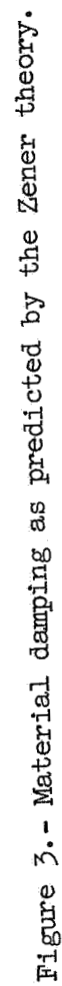
Figure 1.- Photograph showing relative sizes of joint damping models.



SCALE FACTOR $\lambda : 1, .667, .333, .053$

Figure 2.- Dimensions of joint damping models.

$$\gamma = \frac{25 f^2 c}{\pi^2 K}$$



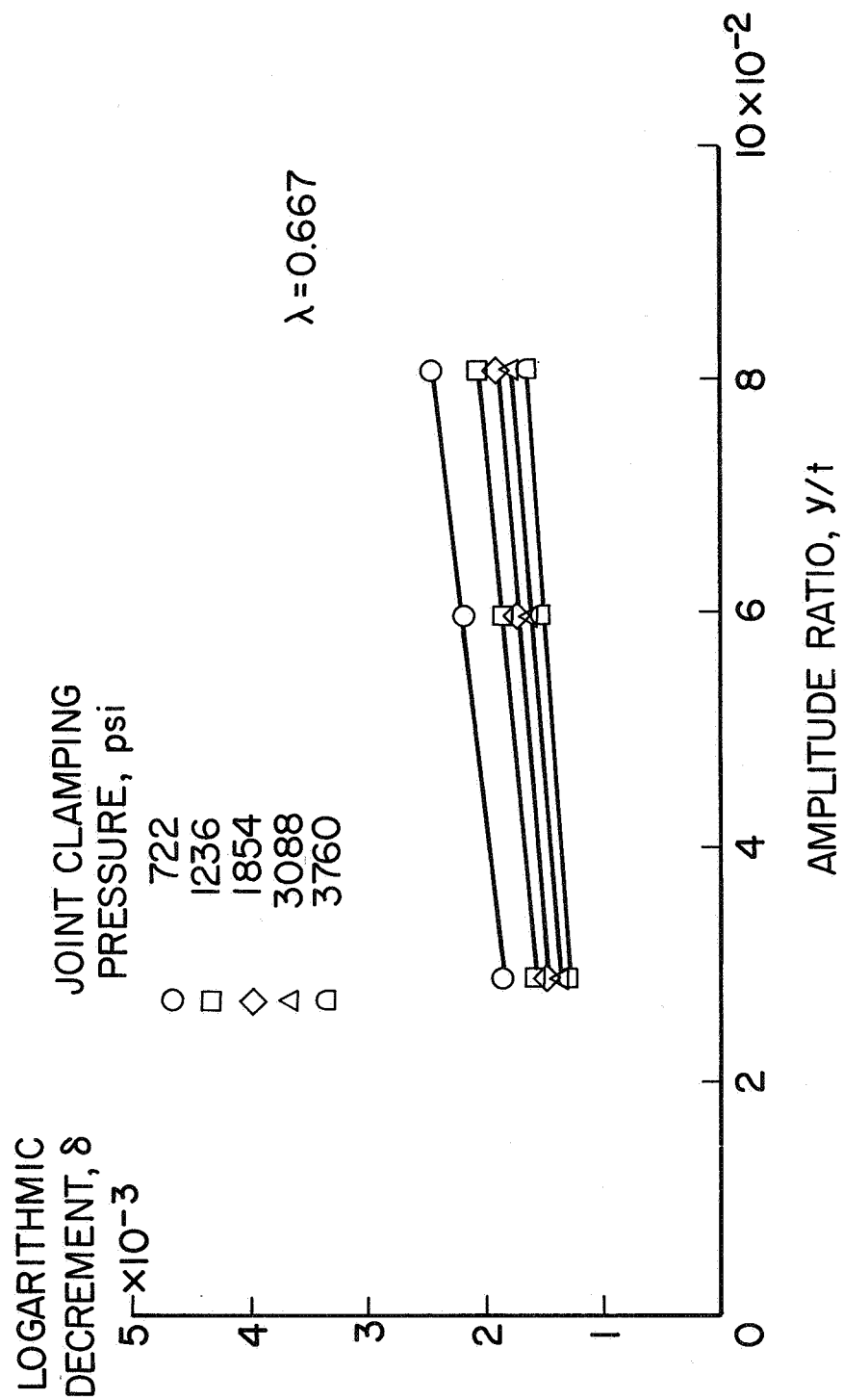
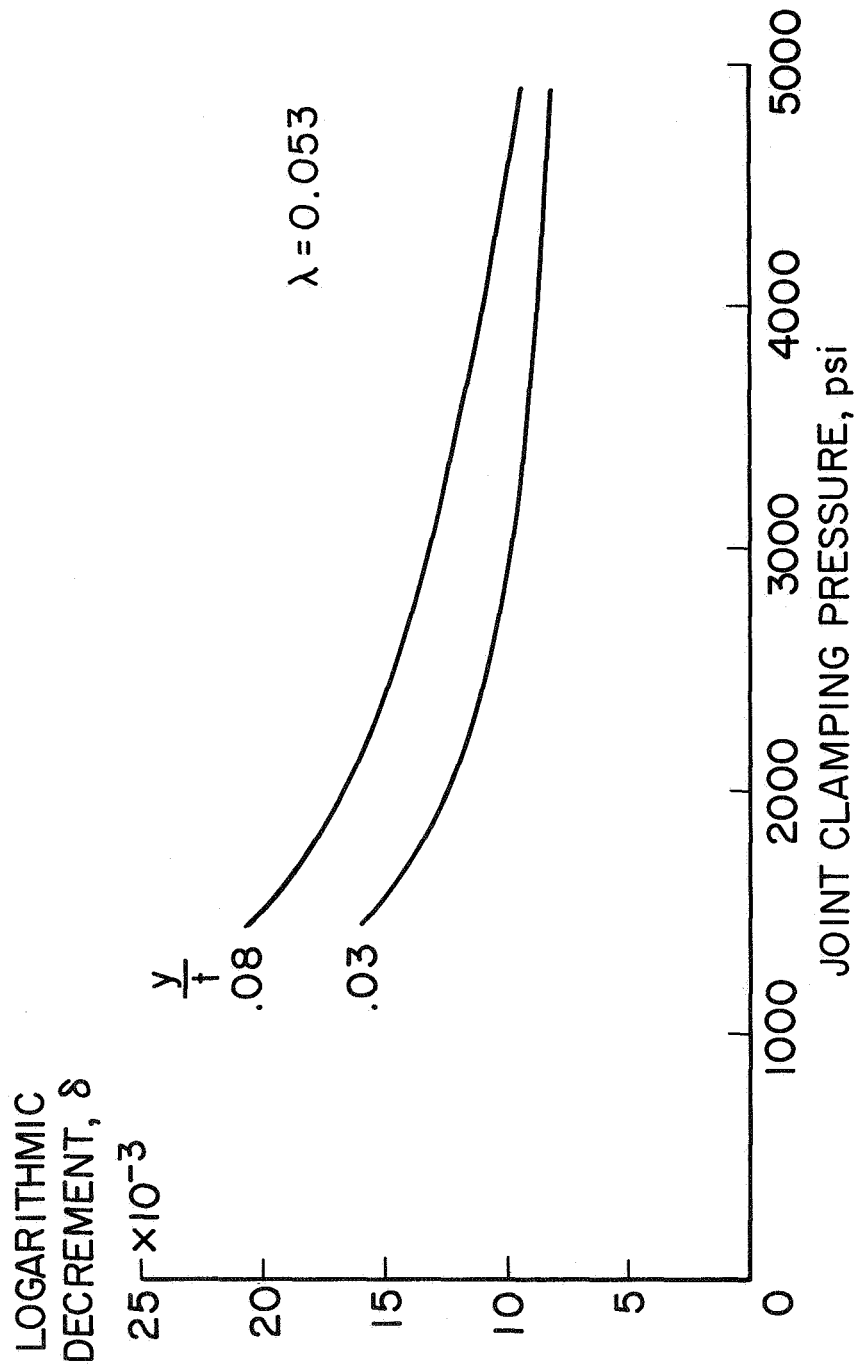
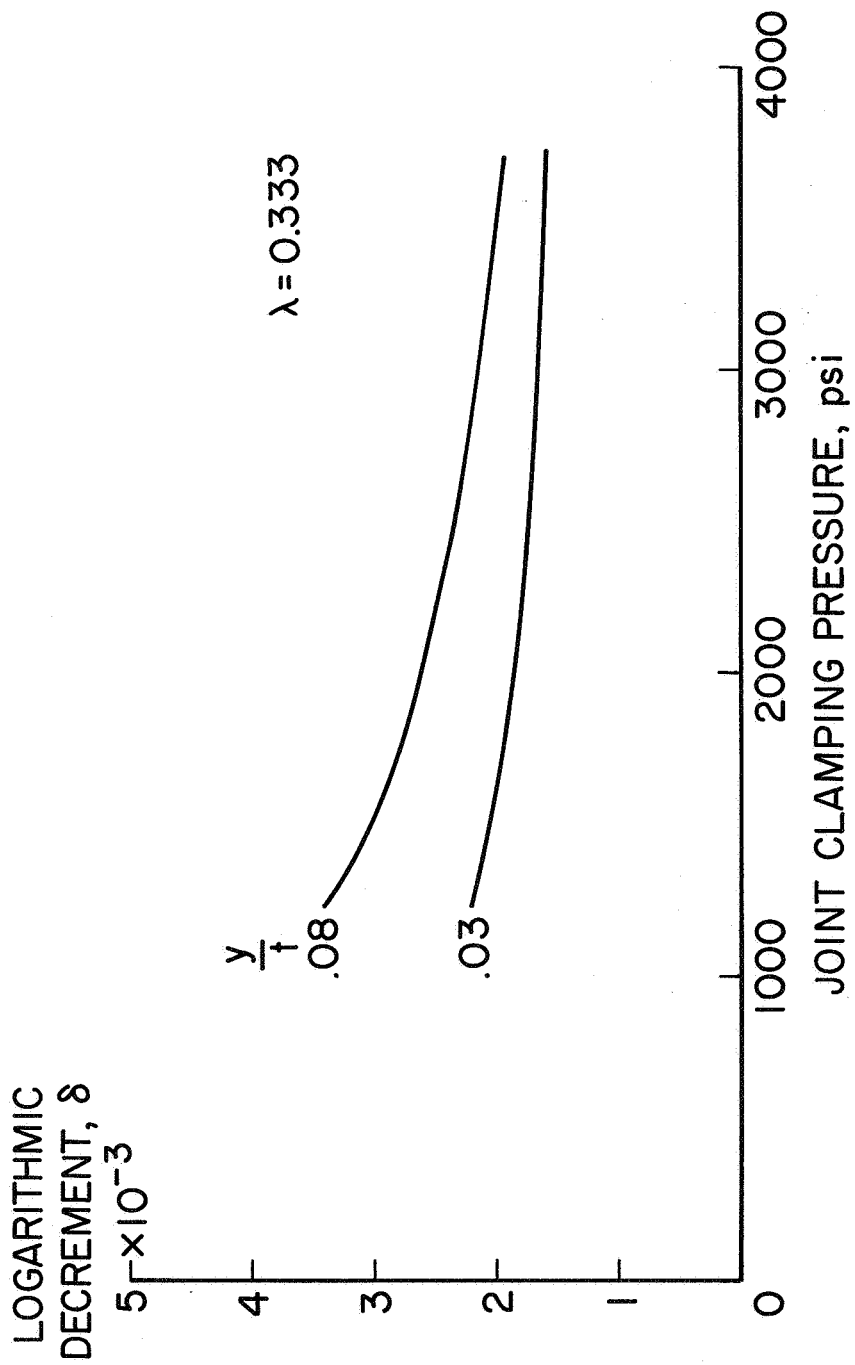


Figure 4.- Variation of total damping with beam-tip vibration amplitude.



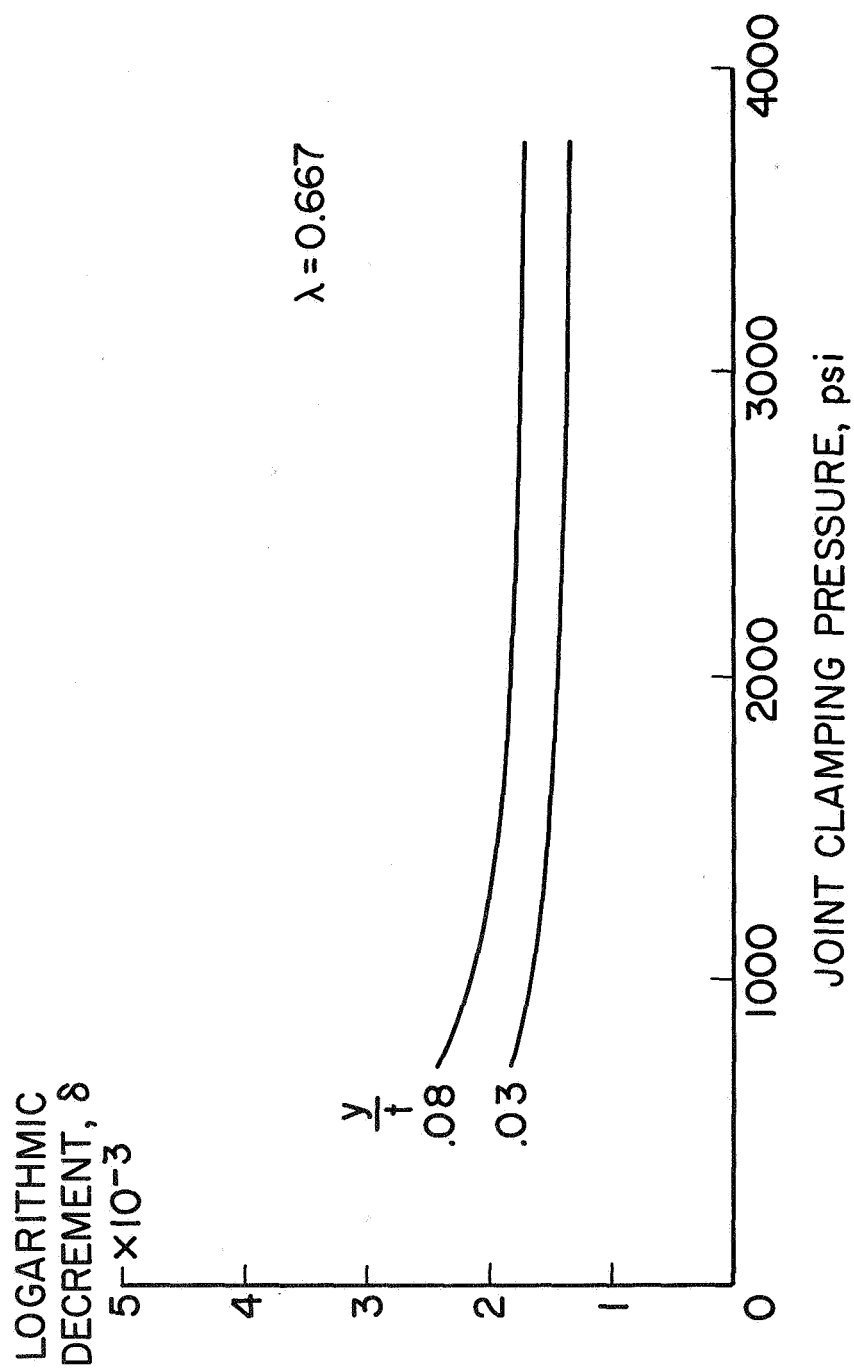
(a) $\lambda = 0.053$.

Figure 5.- Variation of total damping with joint clamping pressure.



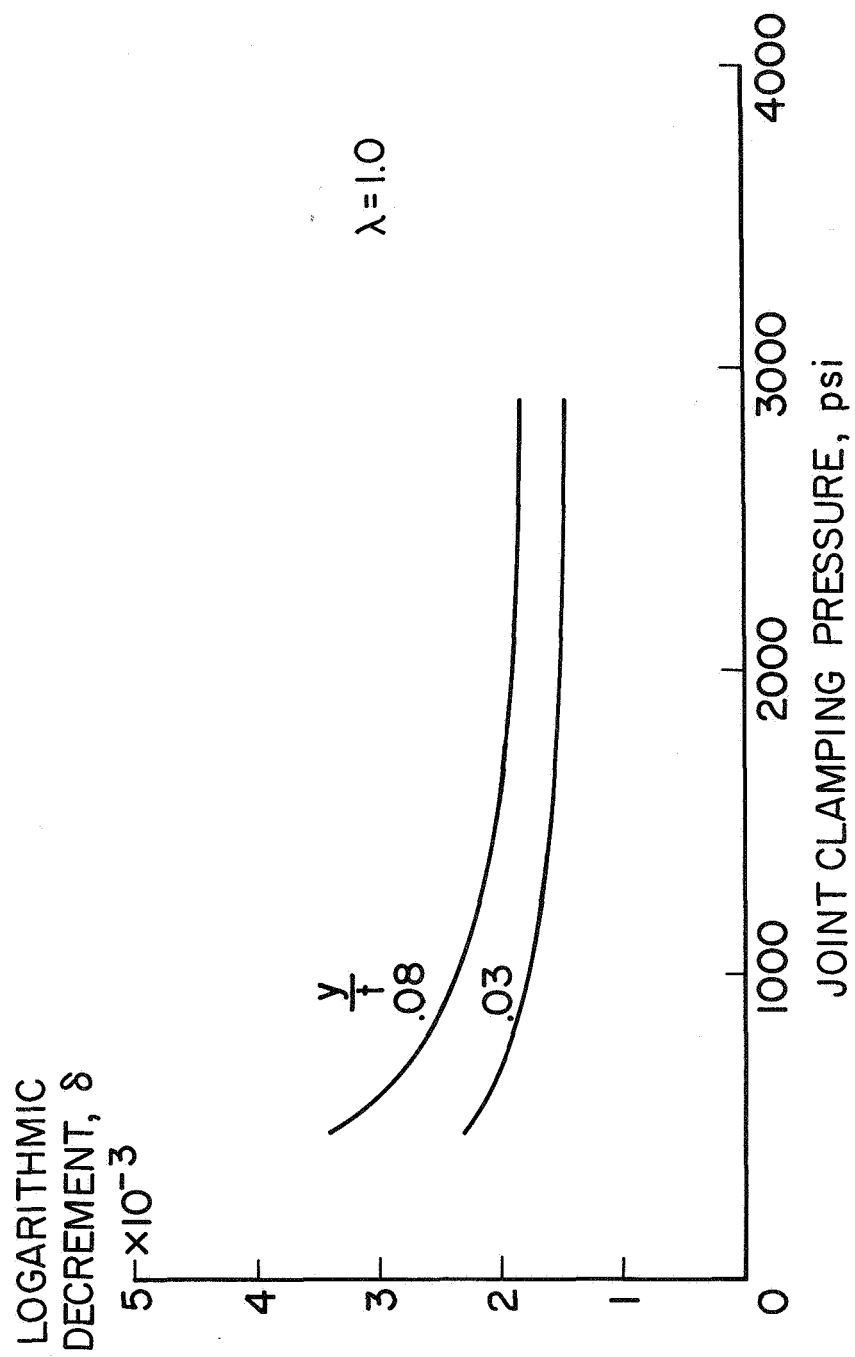
(b) $\lambda = 0.333$.

Figure 5.- Continued.



(c) $\lambda = 0.667$.

Figure 5.- Continued.



(d) $\lambda = 1.0$.

Figure 5.- Concluded.

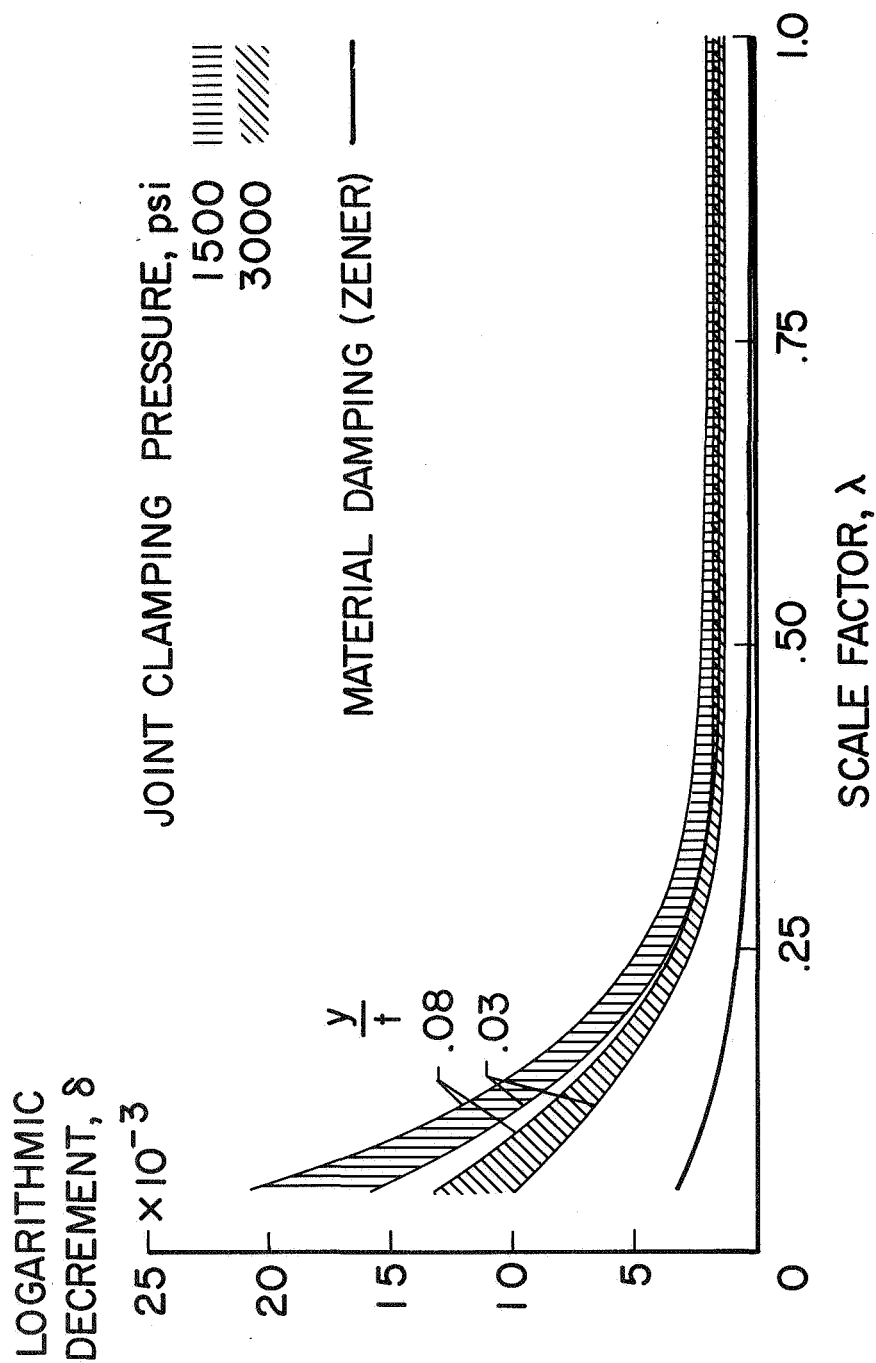


Figure 6.- Variation of total damping with scale factor.

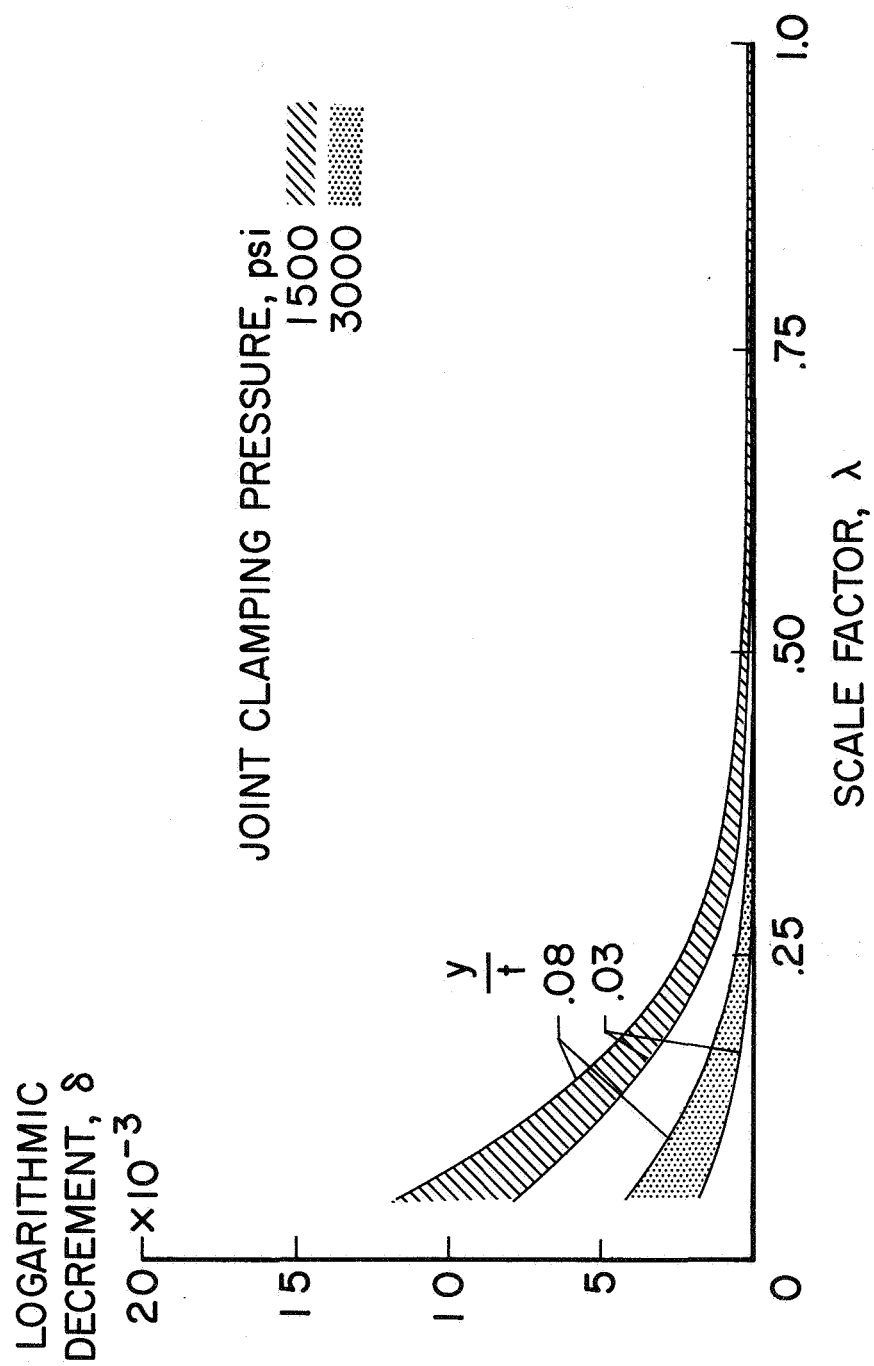


Figure 7.- Variation of joint damping with scale factor.

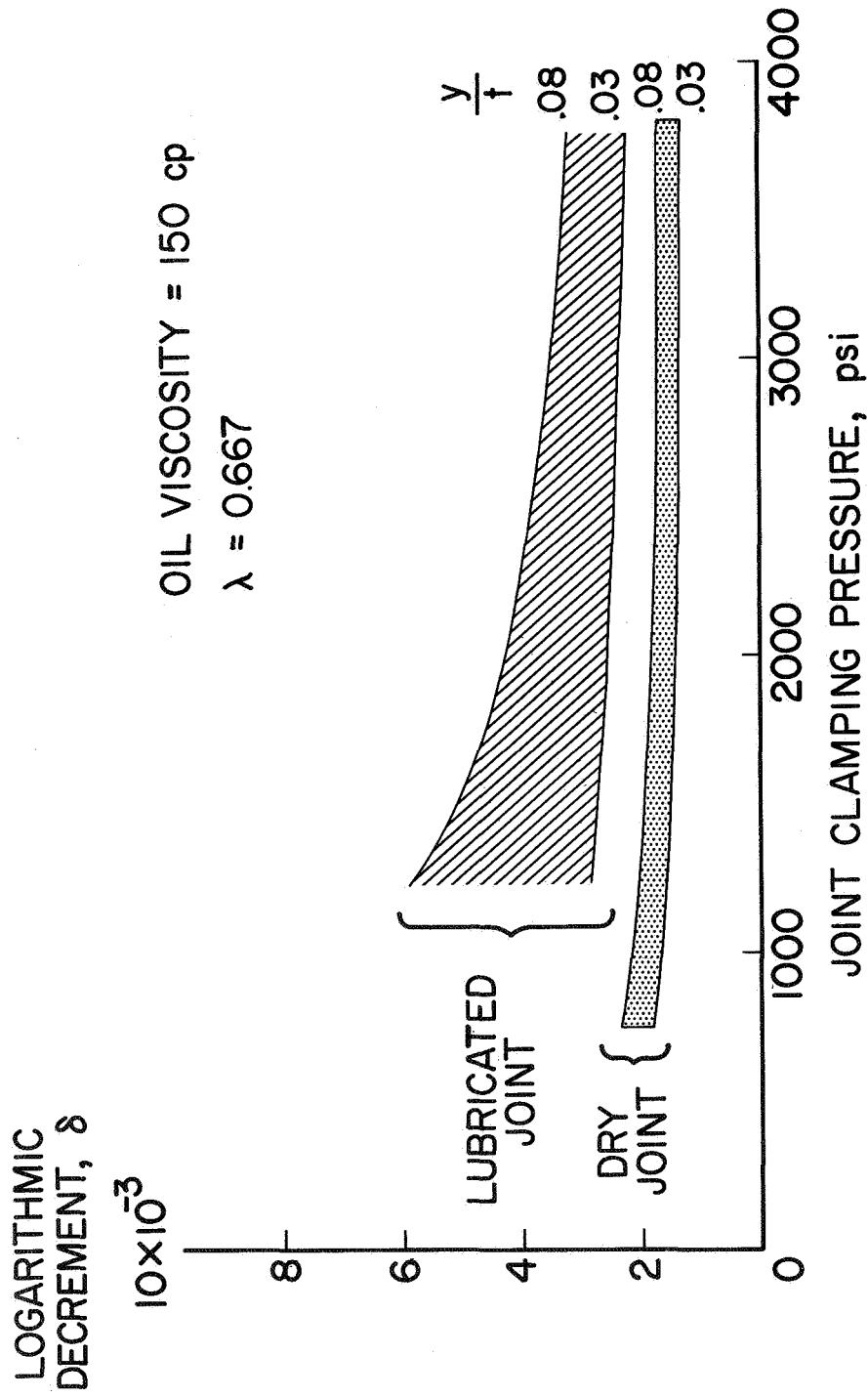


Figure 8.- Variation of total damping with joint clamping pressure for lubricated and dry joint conditions.

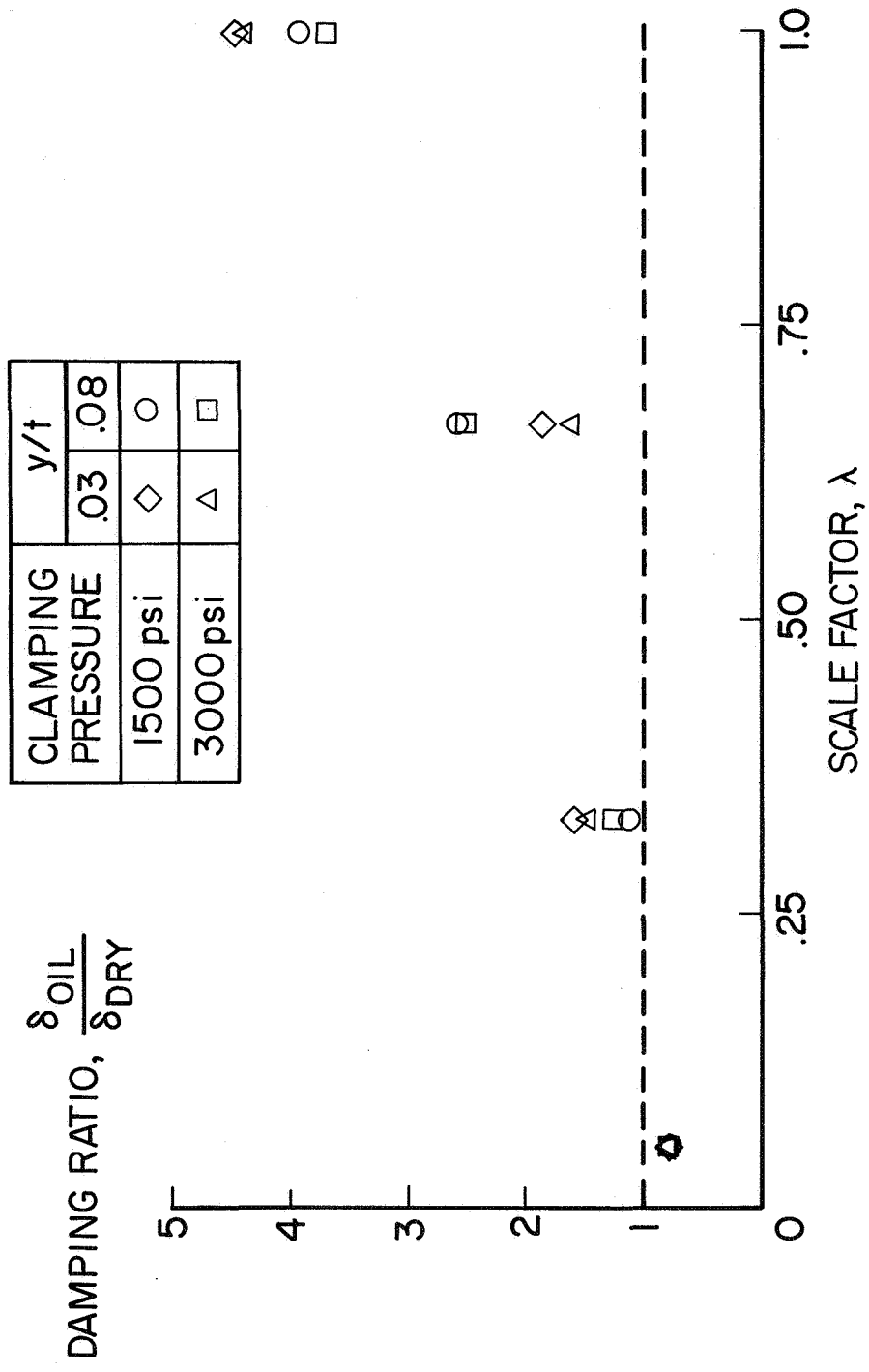


Figure 9.- Joint lubrication effectiveness summarized for all models.

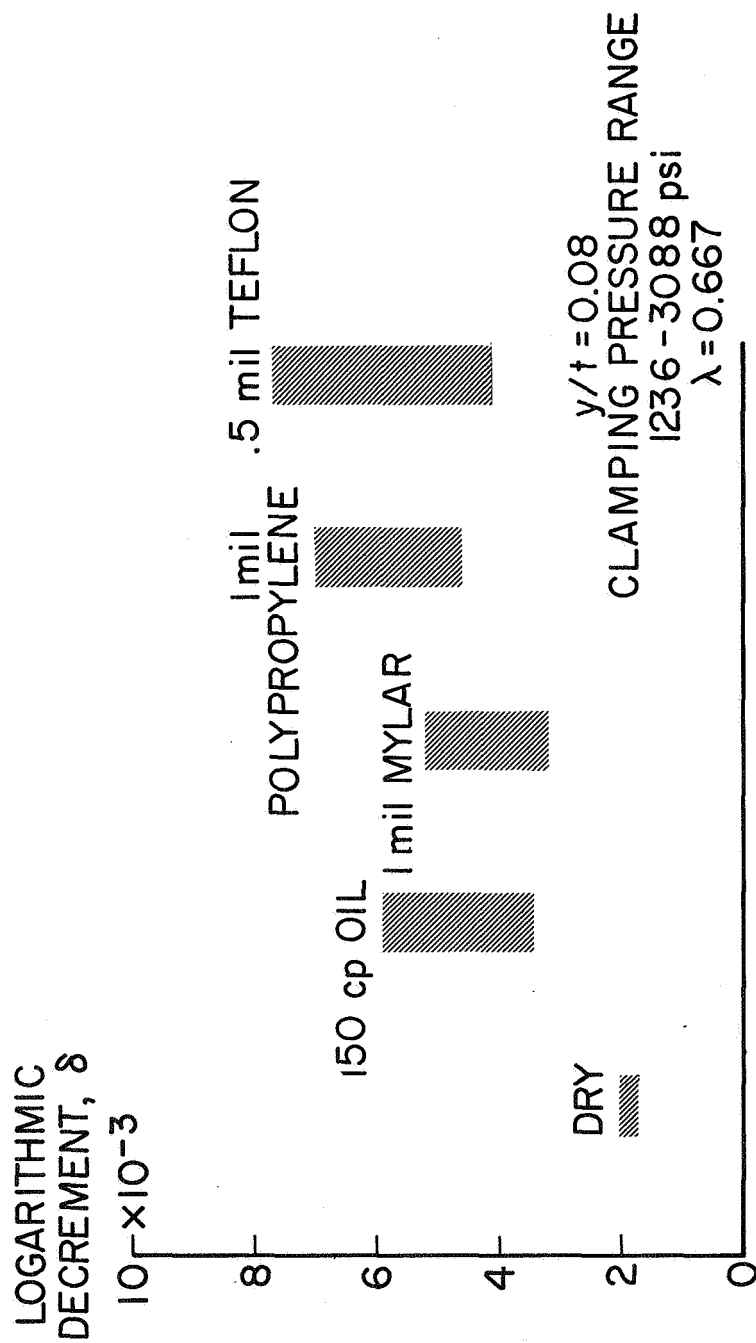


Figure 10.- Total damping ranges obtained for various joint conditions.